A THERMODYNAMIC DESIGN MODEL FOR AUTOMATIC FILM DRYING

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ABSTRACT

The LogE Development and Engineering Staff established that film drying, considered as a fundamental thermodynamic phenomenon was governed by two dynamic processes, heat transfer and mass transfer. Although the mechanisms of the individual rate processes were impractical to define, the envelope within the confines of which these rate processes are accomplished was modeled to develop a realistic design tool relating design air flow and power consumption. This design model was then expanded by the application of "state-of-the-art" concepts in heat exchange and mass flow dynamics to provide a quantitative evaluation of design economies.

BACKGROUND

The introduction of sound on film in the movie industry in the early 1930's brought about the first complete dry-to-dry machine processing of 35mm cine film in roll form. Prior to this advent of a sound track on film, rolls of cine film had been developed, fixed, and washed by machine methods. Drying of the film was accomplished by what would today be regarded as a rather primitive procedure. The processed rolls of film were wiped or squee-geed of the large droplets of water to prevent spotting, and then wound in a spiral fashion around racks and "hung out to dry" in any available space. This method was called "normal air drying" ---- incredible!

We Photographic Scientists today know that film must be dried rapidly, uniformly, and in a dust-free environment. After drying, it must be ultra-sonically cleaned and given a protective coating of optically inert tetrasterate wax. Anything less would be unthinkable, unworkable, and of course, unscientific. It is more than coincidental that all of this technology and science was forced upon us by the fact that non-uniform drying of cine film prevented the acceptance or feasibility of sound-on-film as an entertainment media, or art form.

I would attribute the solution of this problem and the beginning of a new technology to my friend "Hub" Houston, who at the time was working with Howard Hughes (who in turn was working with Jean Harlow-- if you can call that "work"). Besides being a vital part of that golden era of the movie industry "Hub" Houston was a true mechanical genius who had that rare quality of creating a workable, simple and practical solution to a problem for which the tables and formulae had not yet been written. More than thirty years ago "Hub" built the first warm air impingement dryer for 35mm cine roll film. These were the basic problems:

1. The dryer must run at a synchronized speed with the processor;
2. Materials must be found that would not damage the emulsion or base side of the film;
3. The transport mechanism must be smooth running and entirely free of any jerky motion;
4. The film must not be over dried to prevent brittleness; and finally,
5. The ever present fire hazard with nitrate base film must be kept at a minimum.

We have eliminated the nitrate base films and have substituted acetate and polyester base materials. The dilemma of the dynamics of this situation was inevitable. The problems attendant to the drying of 35mm film must be expanded to film widths of 48" and more. The problems do not, however, increase linearly as a function of width, but almost exponentially. The fundamental problems remain the same: 1) width format; 2) drying rates; and 3) economy of operation.

The first step in the chain of events to be applied to wide film drying was to employ the warm air impingement methods used for 35mm films expanded to 9½" aerial roll films. A Thermodynamist would tell you that not only is it necessary to impinge the warm, dry air at both surfaces of the film, but that you must employ means of removing this air immediately to prevent saturation. Also, air velocities sufficient to break up the laminar air layer at the film surface are required. If money were no object, you would impinge upon the film surface not warm air, but dry air preferably at ambient room temperatures and relative humidities ranging from 35% at the dryer entrance to 1% at the center and gradually back to 50%. With available desiccants, such as silica gel and activated alumina, 45KVA of electrical energy is needed on a continuous basis to dry 9½" wide film at 45' per minute. The power requirement is nearly a linear relationship with the drying rate in feet/minute.

In 1958, the co-author of this paper designed and built a "dry air" film dryer for 9½" aerial roll film. The dryer was not connected to a film processor for the reason that the wet section of the processor could not keep up with the dryer capability. Activated alumina was used for the desiccant, because of its dust-less characteristics. Electrical energy or heaters were required to regenerate the desiccant. The dryer cabinet was a series of air conditioning chambers in which the moving film was subjected to various humidity conditions and constant temperature.

This dryer was installed in a downtown Washington Government Agency. Special wiring had to be provided for its power consumption of 45KVA. When the dryer was turned on, there was a noticeable decrease in the speed of the trolley cars operating outside. The trolleys were very shortly replaced with buses. I mention this only to illustrate what is possibly the most important part of Mr. Harter's portion of this paper, that the true solution of film drying must be based on the optimizing and the economizing of a particular device in terms of the known parameters.

In conclusion, let me again quote Mr. Houston, "For every problem there are many solutions; one of these solutions will be elegant in its simplicity".

INTRODUCTION

The design procedures followed in the development of automatic film processors are, in many areas, quite empirical. This is particularly so for the dryer subassemblies of the film processors. This empirical approach to design can be most directly related to the variability in film backing and emulsion characteristics, both with one manufacturer and between manufacturers. However, film drying is a thermodynamic process. As such, even though this process may not be exactly definable, the process envelope within which a dryer must operate can be defined and modeled. The general thermodynamic model of this process envelope then becomes the confines within which the design mechanism must operate.

The object of this presentation is to describe how we, at LogE, established this basic thermodynamic model, then used this model to develop areas in the general design procedure which allowed both economical and operational optimization in design.

DISCUSSION

A. Basic Thermodynamic Process

Fundamentally, a film dryer incorporates two dynamic processes, a heat transfer process and a mass transfer process. Obviously these processes are directly related, each being partially defined and limited by the other. In particular a conveying means, usually air, is required to be brought in intimate contact with a wetted film surface, acquire the moisture from this surface and then convey it away. This is the basic mass transfer process. However, in order to facilitate this process, there must be an energy transfer. First, energy must be transferred from some source to the conveying air stream so that this conveying stream has the energy potential to (a) "pick-up" the moisture from the film surface and (b) transport this moisture away from the film. Second, this energy must be partially transferred to the surface moisture so that it will evaporate and diffuse into the conveying air stream. This is the basic heat transfer process. The process model for this system is shown on Figure 1.

There is a mutual exchange of heat and moisture in the process transfer zone which, from a definition standpoint, is very complex. The complexity of the process behavior in this area is related to the difficulty of defining the heat transfer and mass diffusion coefficients which are uniquely peculiar to each individual configuration.

From a design standpoint, accommodating this complexity is not necessary, however, since we can base design parameters on net overall material and energy balances. The fundamental equations which model these balances are also shown on Figure 1.

The problem, from a design standpoint, becomes to relate the air flow and power consumption required to achieve a prescribed water removal capability. Then, with that relation, study the controlling parameters and establish what optimization and economies can be effected by manipulating operation within the confines of the developed model.

Obviously, modeling this thermodynamic system will have to be predicated on some psychrometric formulation. There is not an exact psychrometric formula and the approximation that can be used is extremely difficult to handle from the standpoint of formulating continuous functions. Consequently the model, as developed in the following pages, is based on specific end conditions and point by point design relations are derived which relate water removal, power, airflow, and end conditions. The trends characterized by these specific points are then presented in the form of design curves.

B. Model Development

Figure 1 schematically represents the drying system. For this part of the discussion the heat exchanger shown between points 1 and 2 is not considered so conditions at these points are identical. The psychrometric description of the same system is shown on Figure 2. Note here that specific end conditions are chosen for points 2 and 3 and also that the psychrometric process to be followed between points 3 and 4 is defined.

At point 2 the process starts with what is considered normal laboratory room conditions 70° and 50% RH. The process air is then heated at constant specific humidity to 130° F. This 130° point was chosen as a maximum since it is felt that temperatures in excess of this could easily lead to film surface damage. Next, in passing through the process transfer zone, the air follows an adiabatic saturation process to achieve some degree of saturation.

At this point, it is important to establish that the process assumptions are understood and reasonable. First, consider the assumption that adiabatic saturation is the mechanism describing the mass transfer. It is realized that in any engineered system there are losses, likewise in ours. However, there is no other mechanism which could be assumed to describe more accurately what is happening at this point. Testing verified this assumption to be practical. Next, it is not known and cannot be predicted just what degree of saturation is reached at point 4. The logical approach is to assume various end points for condition 4 and see what effect this variation has on airflow and power consumption. Consequently, as shown on Figure 2, the various end points 4_1, 4_2, 4_3 and 4_4 are designated.

The following table lists the important design parameters describing these various process points:

POINT CONDITION	TEMP DB-F°	RH-%	SPECIFIC HUMIDITY GR/#AIR	ENTHALPY BTU/#AIR
2	70	50	55	25.4
3	130	8	55	40.4
4_1	118	15	74	40.4
4_2	110	22	87	40.4
4_3	103	30	98	40.4
4_4	92	50	116	40.4

These psychrometric conditions must first be related to the "general" film drying requirement to be met. Then the affect varying these conditions will have on film drying and film drying economics is analyzed.

Water removal is the primary engineering objective and must first be defined. The water supply can be formulated as is shown on Figure 1. This is the mass that has to be removed during drying:

Water Rate, $W = A_1 \times v \times \rho$ in Pounds of Water/minute

Where A = Film width in feet
v = Transport velocity in fpm
ρ = Water density on film in pounds/feet^2

Unfortunately the ρ item is this expression is an unpredictable variable almost impossible to characterize by trends. Consequently the developed model is built on a "Per pound of water removed per hour" basis.

The energy requirements which will now be developed are related to the theoretically minimum amount of energy required to evaporate a pound of water. This latent heat of evaporation which, for our specified conditions, is the difference between the enthalpies of saturated steam at 130° F and saturated liquid at 70° F, amounts to 1080 BTUs per pound of water. This is the amount of energy that each pound of water <u>must</u> pick up from the air stream in order to demonstrate a change in state. To do this however, an amount of energy must be supplied which is directly related to the useful psychrometric carrying capacity of our conveying air stream. This energy is determined as follows:

The minimum amount of air required to pick up one pound of water is:

$$M_{air} = \frac{1\#H_2O \times 7000\ GR/\#}{(GR_{air\ 4} - GR_{air\ 3})} = \frac{7000}{GR_{3-4}}\ \#\ Air/\#H_2O\ Evap.$$

Note here that varying conditions are assumed for $\Delta\ GR_{3-4}$ and a trend for this variation is to be established.

At this point in the analysis it is convenient to convert this mass flow of air per pound of water evaporated to standard volume flows based on a "one pound of water evaporated per hour" rate.

$$SCFM = \frac{M}{60 \times .075\ \#/SCF} = \frac{M}{4.5}\quad SCFM/\#H_2O\ Evap/Hr$$

The energy requirement that is <u>to be supplied</u> to this mass flow is dependent on the established enthalpy change between conditions 2 and 3 and amounts to:

$$E = M(h_3 - h_2)\quad BTU/\#H_2O\ Evaporated$$

The following table presents the results of calculations made as outlined above for the modeled system. The results are characterized separately for each of the four conditions assumed at point 4.

CONDITION AT POINT 4	BTU SUPPLIED PER $\#H_2O$ EVAP.	WATTS PER $\#H_2O$ EVAP/HR	SCFM PER$\#H_2O$ EVAP/HR
118° F	5520	1620	81.8
110	3290	916	48.7
103	2440	713	36.2
92	1730	505	25.5
Latent Enthalpy	1080	315	

Curves A and B in Figure 3 present these data in graphic form. The significance of the trend established by varying the conditions at point 4 is put into more realistic perspective when curve A is analyzed with respect to curve B as a reference.

C. Observations Affecting Performance and Economy

Assuming satisfactory performance the cost of operation is, obviously, directly related to the "thru put" quantity of air moved and heated. From Figure 3, the specific performance model, it can be seen that this cost is then directly, but not linearly, related to the discharge dry bulb temperature. Therefore, to optimize performance and economy of operation one of two things must be done. Either provision must be made to discharge at a lower condition, i.e., dry bulb temperature, or something must be done to take advantage of the high discharge condition and put it to work.

The first approach is the most directly practical. To establish what must be done to lower the dry bulb it is necessary to define why the discharge condition is high. Obviously, lack of mutual heat and mass transfer. To improve this situation three process parameters have to be improved: heat transfer coefficients, diffusion coefficients and residence. These all can be improved if, in the process transfer zone, mass flows and mass velocities are increased. This then is what should be done; the process air stream should be recirculated in sufficient quantities and at sufficient velocities within the process transfer zone so that the much smaller throughput effluent attains the discharge temperature (and RH) that is desired. The recirculation amount has to be determined empirically through developmental testing of each particular configuration. But, and this is significant, the desired discharge condition can be met.

Another point of significance here is that no matter how much of a process mass is recirculated the input heat (a measure of operating cost) is supplied only to the smaller throughput quantity.

The other technique of optimizing performance is to take advantage of, and use, the high discharge temperature. This is done every day in a great many processes by economizing through heat transfer surface. Figure 1 shows schematically how heat exchange can be introduced into the film drying process.

The compact recuperative heat exchange formulations developed by McAdams[1] and expanded by Kays & London[2] can be used to give a close approximation of the net improvement in process performance by relating thermodynamic efficiency to relative temperature profiles. The formulation for this improved performance, as developed in the appendix to this presentation, is:

$$\mathcal{E} = \frac{\text{Energy Supplied}}{\text{Energy Required}} = \frac{t_3 - \left\{ t_1 + \epsilon\,(t_4 - t_1) \right\}}{t_3 - t_1}$$

which for our specific set of boundary conditions resolves down to:

$$\mathcal{E} = 1.84 - \frac{t_4}{86}$$

When using $t_3 = 130^\circ$ F.,

$t_1 = 70^\circ$ F.,

and $\epsilon = 0.70$ effectiveness of heat transfer.

The impact of this economizing through heat transfer is shown graphically on Figure 3, as a function of discharge temperature and in relation to the latent enthalpy requirement. It should be pointed out that the improvement reflected by this analysis is based purely on sensible heat transfer. In reality, however, the discharging effluent also has a latent energy content of reasonable proportion which will reflect itself through the heat transfer as a slight improvement over that shown.

APPLICATION OF TYPICAL PROBLEM

The following development and analysis of a typical design problem is presented to demonstrate application of the described data. Ambient operating conditions of 70° DB and 50% RH are used. Future, more extensive design work would, however, require development of a family of design curves similar to that shown as Figure 3 so that a practical range of ambient conditions could be accommodated by the complete design model.

Consider first that the materials which a proposed design would have to process should be as wide (w) as 42 inches (3.5') and have a maximum surface entrainment (ς) of 0.01#/ft.^{2}. The machine is required to process this film at a velocity (v) of 5' per minute maximum. These conditions therefore dictate a maximum water removal requirement (W) of:

$$W = W \times v \times 60 \times \varsigma \quad \#H_2O/Hr$$

$$= 3.5 \times 5 \times 60 \times .01$$

$$W = 10.5 \ \#H_2O/Hr$$

The dryer design, as stated previously, would be based on using a 70° DB and 50% RH ambient, heating this air at a constant humidity to 130° F and then adiabatically removing the surface moisture from the film until a design effluent of 118° DB and 15% RH is reached.

Referring to Figure 3, we make the following determinations for power (P) consumption and airflow (SCFM):

$$P = 1.62 \ \frac{KW}{\#H_2O/Hr} \times 10.5 \ \#H_2O/Hr$$

$$= 17 \ KW$$

&

$$SCFM = 81.8 \ \frac{SCFM}{\#H_2O/Hr} \times 10.5 \ \#H_2O/Hr = 860 \ SCFM$$

The power required to achieve this same water removal capacity when employing 0.70 effective heat exchange would be only:

$$P_{\Delta XGR} = 0.713 \ \frac{KW}{\#H_2O/Hr} \times 10.5 \ \#H_2O/Hr$$

$$= 7.5 \ KW$$

or a 56% power reduction.

This same 56% reduction in power can also be achieved by increasing the mass flow recirculation within the process transfer zone sufficiently to maintain 368 SCFM (i.e., 35 x 10.5) through put having a 103° DB and 30% RH effluent condition.

The determination of incorporating either the 0.70 effective heat exchanger or an additional internal air flow recirculation system is dependent entirely on the economies of the design configuration. Both approaches will achieve the same design goal; the least expensive would be selected.

CONCLUSION

The preceding development has shown that by employing the "Engineering Method", a model of the operating envelope for the controlling film drying parameters could be developed. This model provides considerable insight into the factors governing the economics of film drying.

4/30/68
RH;EK:jhs

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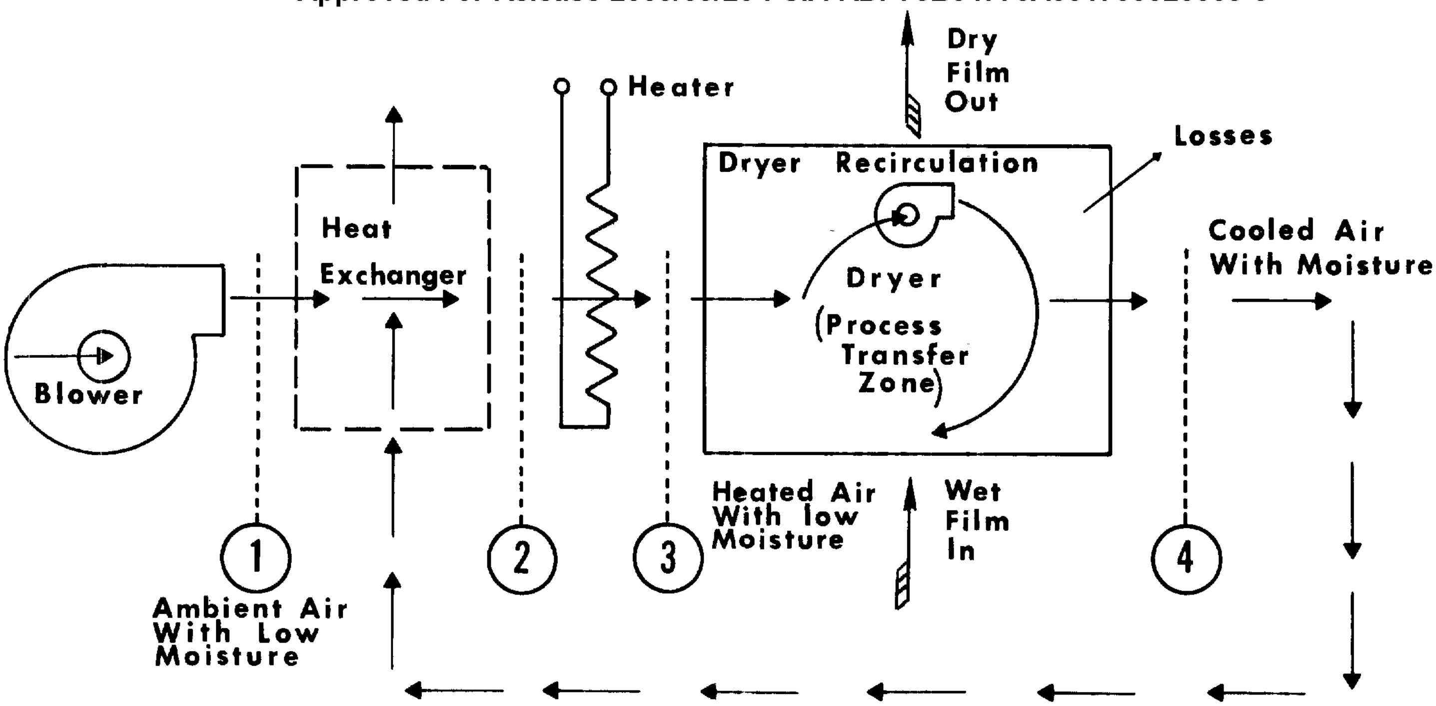


SYSTEM FLOW DIAGRAM

MOISTURE MASS BALANCE

MOISTURE IN = MOISTURE OUT

$$A_{\substack{FILM\\AREA}} \times v_{FPM} \times \rho_{\#H_2O/FT^2} = Q_{\substack{CFM\\AIR}} \times m_{\#AIR/FT^3} \times (\omega_{4_{OUT}} - \omega_{3_{IN}}) \ \#H_2O/\#AIR$$

ENERGY BALANCE

ENERGY IN = ENERGY OUT

$$\left(Q_{\substack{CFM\\AIR}} \times m_{\#AIR/FT^3} \times h_{3\ BTU/\#AIR}\right) + A_{\substack{FILM\\AREA}} \times v_{FPM} \times \rho_{\#H_2O/FT^2} \times h_{f\ BTU/\#H_2O} = \left(Q_{\substack{CFM\\AIR}} \times m_{\#AIR/FT^3} \times h_{4\ BTU/\#AIR}\right) + LOSSES$$

FIGURE 1

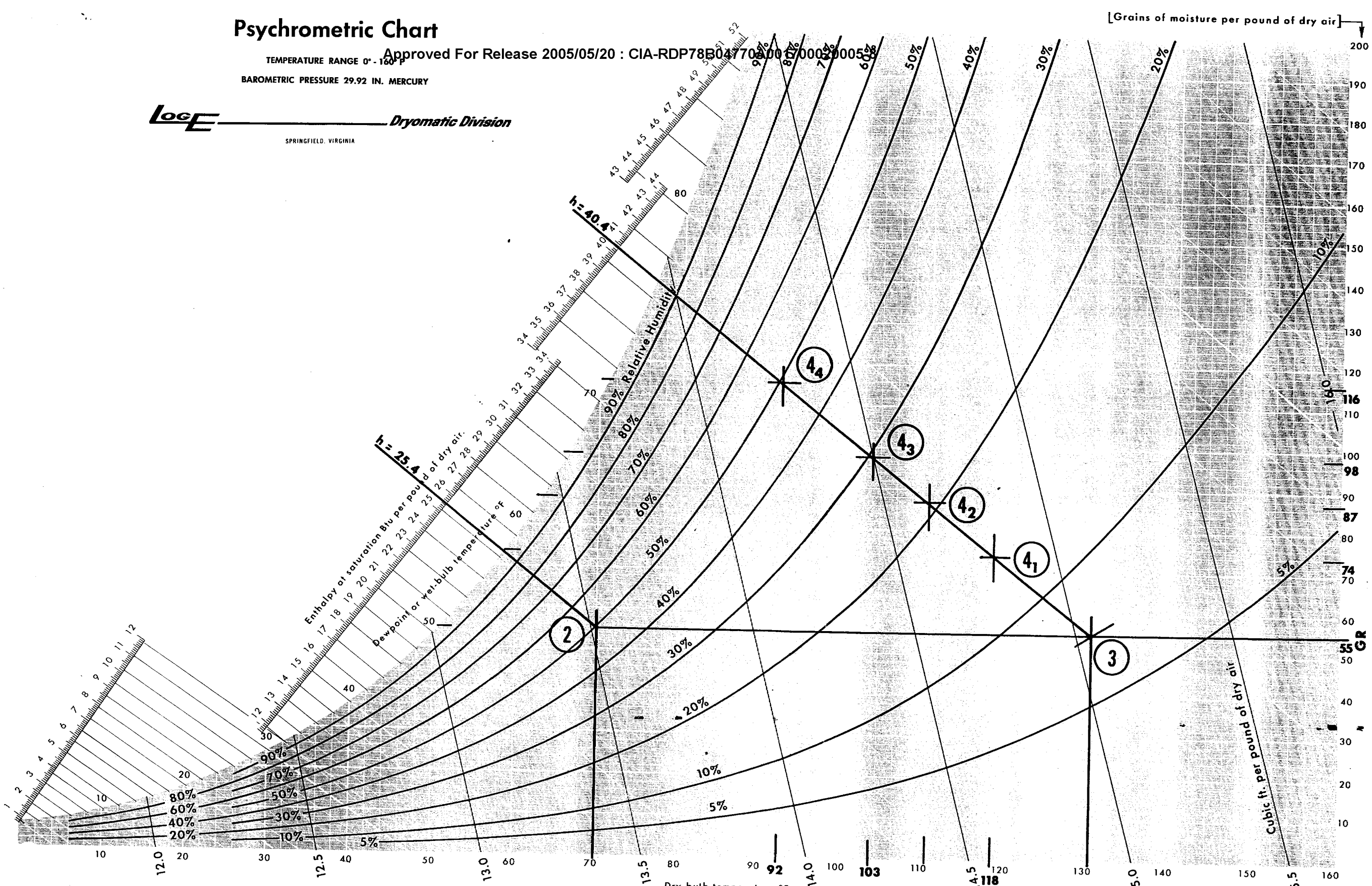
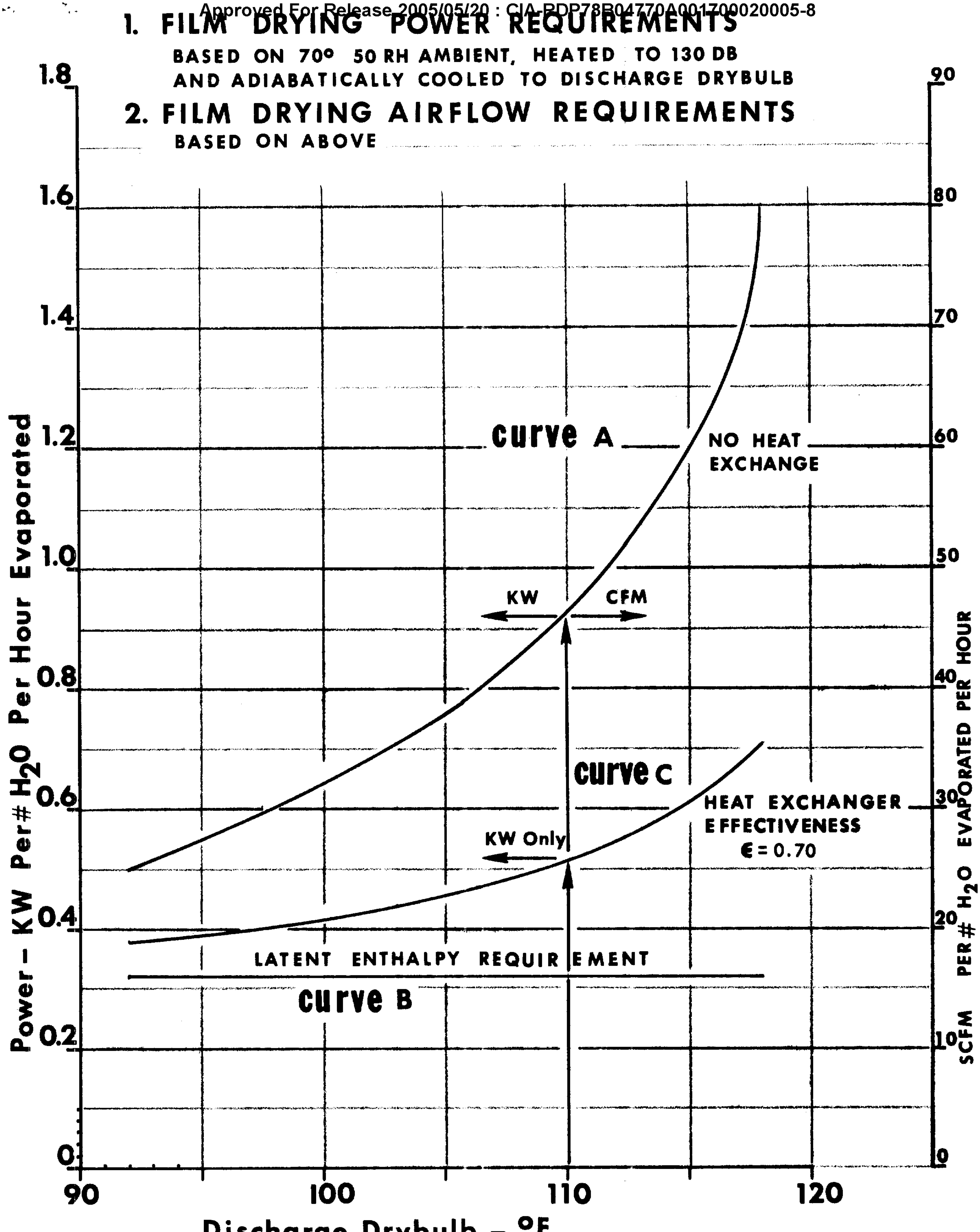

Psychrometric Chart
TEMPERATURE RANGE 0° - 160° F
BAROMETRIC PRESSURE 29.92 IN. MERCURY
LogE Dryomatic Division
SPRINGFIELD, VIRGINIA
Grains of moisture per pound of dry air
Enthalpy at saturation Btu per pound of dry air
Dewpoint or wet-bulb temperature °F
Relative Humidity
Cubic ft. per pound of dry air
Dry-bulb temperature °F
h = 25.4
h = 40.4
GR
PRINTED IN U. S. A. 3-65

FIGURE 2



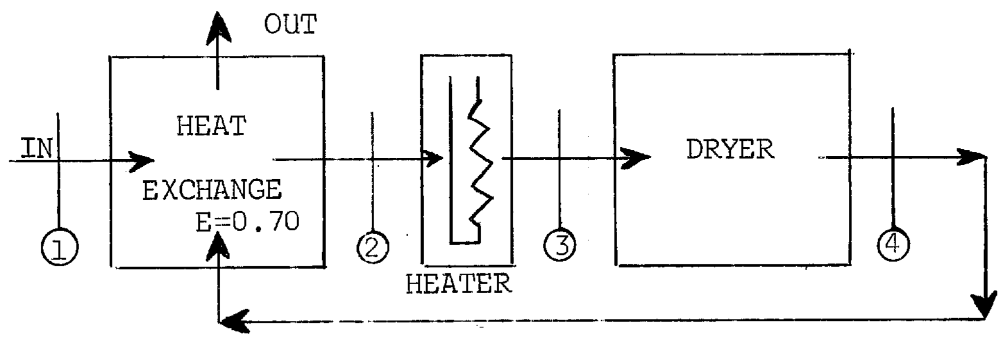
1. FILM DRYING POWER REQUIREMENTS
BASED ON 70° 50 RH AMBIENT, HEATED TO 130 DB
AND ADIABATICALLY COOLED TO DISCHARGE DRYBULB
2. FILM DRYING AIRFLOW REQUIREMENTS
BASED ON ABOVE
Power - KW Per # H_2O Per Hour Evaporated
SCFM PER # H_2O EVAPORATED PER HOUR
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0
90
80
70
60
50
40
30
20
10
0
curve A
NO HEAT
EXCHANGE
KW
CFM
curve C
HEAT EXCHANGER
EFFECTIVENESS
$\epsilon = 0.70$
KW Only
LATENT ENTHALPY REQUIREMENT
curve B
90
100
110
120
Discharge Drybulb – °F

FIGURE 3

APPENDIX

EFFECT OF HEAT EXCHANGE ON POWER DEMAND:

SYSTEM SCHEMATIC



1. The amount of heat required for drying is:

$$Q_{\substack{\text{Heat}\\\text{Required}}} = mc\,\Delta t = \rho\,(\text{CFM})\,C_p\,(t_3 - t_1) = K(t_3 - t_1)$$

2. But the heat supplied is:

$$Q_{\text{supplied}} = mc\,\Delta t = \rho\,(\text{CFM})\,C_p\,(t_3 - t_2) = K\,(t_3 - t_2)$$

3. These energy quantities are related by end-point temperatures thru heat exchange as:

$$\epsilon = \frac{t_2 - t_1}{t_4 - t_1} \quad \text{or } t_2 = t_1 + \epsilon\,(t_4 - t_1)$$

4. Defining the effectiveness (efficiency) of our Power heat system, i.e. $Q_{\text{supplied}}/Q_{\text{required}} = \mathcal{E}$

We have:

$$\mathcal{E} = \frac{Q_{\text{supplied}}}{Q_{\text{required}}}$$

$$\mathcal{E} = \frac{t_3 - t_2}{t_3 - t_1} = \frac{t_3 - t_2}{t_3 - t_1}$$

But:

$$t_2 = t_1 + \epsilon\,(t_4 - t_1)$$

Therefore, our general solution becomes:

$$\mathcal{E} = \frac{t_3 - (t_1 + \epsilon (t_4 - t_1))}{t_3 - t_1}$$

5. Adapting this general solution to our specific conditions and using values of:

$$t_3 = 130^\circ \text{ F}$$
$$t_1 = 70^\circ \text{ F}$$
$$\epsilon = 0.70$$

We get:

$$\mathcal{E} = \frac{130 - \{70 + .7 (t_4 - 70)\}}{(130 - 70)}$$

And finally:

$$\mathcal{E} = 1.82 - \frac{t_4}{86}$$

4/30/68
RH;EK:jhs